REEVE'S ENTROPY-TEMPERATURE ANALYSIS APPLIED TO THE PORTER-ALLEN ENGINE FOR THE DETERMINATION OF THE VARIOUS HEAT LOSSES

CHARLES C. DAVIS
THOMAS C. CRAIG

1898

TOWNE
378.748
OS 1898.2
REFERENCE

LIBRARY
UNIVERSITY of
PENNSYLVANIA

Rittenhouse Orrery

For Reference

Not to be taken from this room

2537
THESIS
990

Reeve's Entropy-Temperature Analysis Applied to the Porter-Allen Engine For the Determination of the Various Heat Losses.

A Thesis

by

Charles C. Davis

Thomas C. Craig

Submitted May 28, 1898.

Porter-Allen Engine Heat Losses. Davis & Craig '98.

82

Towne
378.748
POS 1898.2

UNIVERSITY
of
PENNSYLVANIA
LIBRARIES

The object of this series of tests was to determine the distribution of the various losses of heat in the engine and also to determine the effect of cylinder covering and of different cut-offs, steam-pressures and amounts of compression etc. on these losses.

The value of these tests is evidenced by the fact that both the steam and exhaust valves of the engine are worn and very leaky. The atmospheric valves are provided with adjustable pressure limits but were worn [illegible] up so tightly as to cause excessive friction and interfere with the action of the governor. Considerable steam leaks through. This causes the water consumption per H.P. hour to be excessively high and greatly modifies the value of some of the results.

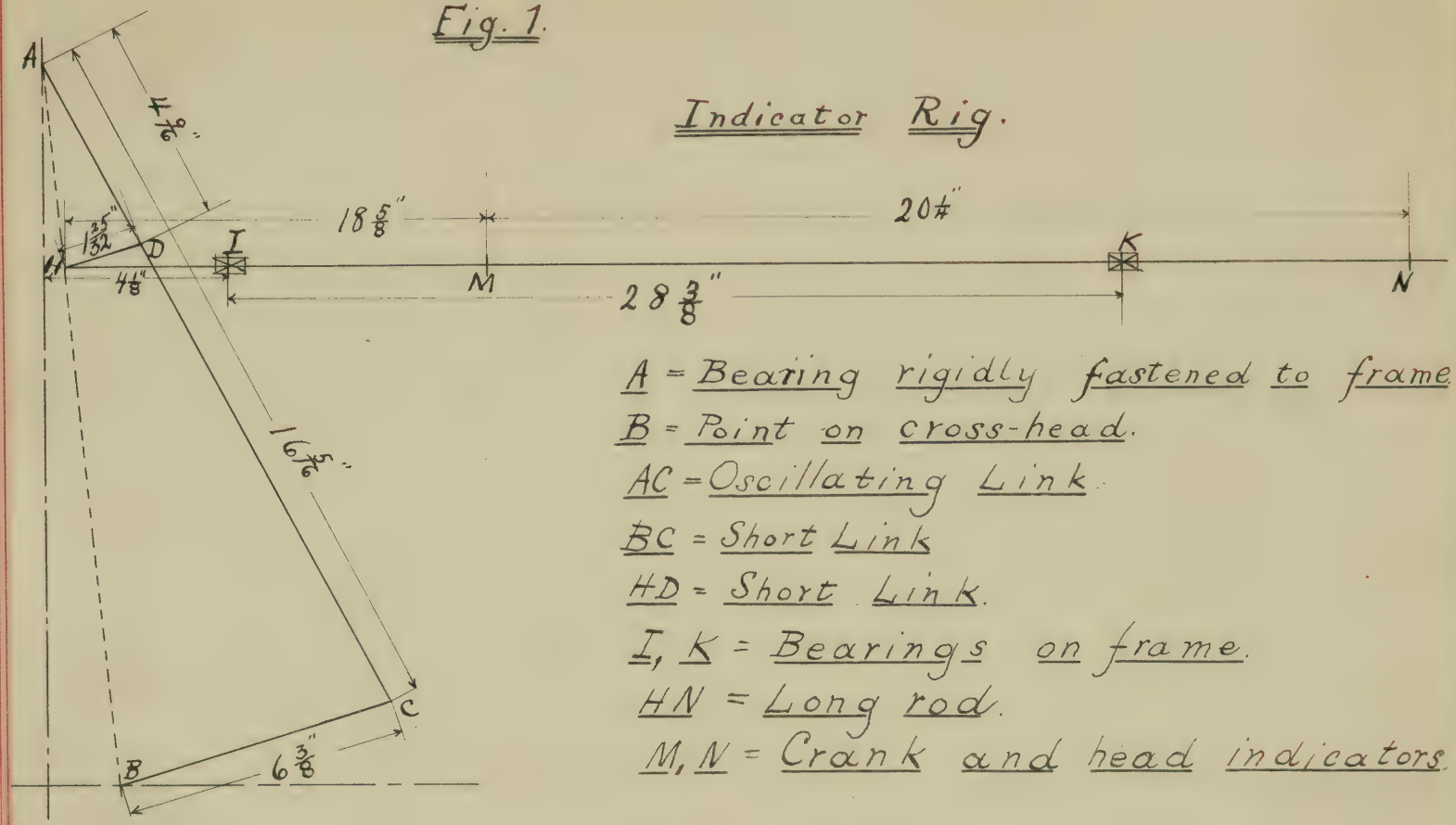
The application of the Stanley — temperature recording instruments accurate indicator cards as the basis from which to start. The indicator-rig previously on the engine consisted of an oscillating link and sector, the indicator-drum taking its motion from the periphery of the sector. This gives a motion which is only approximately true, and is not accurate enough for our purpose. It would cause error in the cards in the cord length of string required for this rig.

In order to eliminate as much as possible this latter error and also to get a mathematically perfect motion we devised the indicator-rig represented in Fig. 1, page 5. This is very convenient to work with and has given satisfactory results.

The conditions that determined the design are as follows. The stroke of the engine is 16 inches. The cards for the sake of accuracy should be as long as possible; but the maximum length of card obtainable with the Improved Thomson Indicators, which we used is $4\frac{3}{8}$ inches. Therefore the ratio of the arms of the oscillating links should be $\frac{4.375}{16}$. On the ground of economy it was advisable to use the old link BC and the bearing at A. The length of the link AC is $16\frac{5}{16}$ inches and therefore the length of AD should be $\frac{4.375}{16} \times 16\frac{5}{16} = 4\frac{9}{16}$ inches. The length of the short link HD is now determined by the fact that to give a correct motion it must always be parallel to BC and H must be in the line AB i.e. ADH and ACB are similar triangles.

Fig. 1.

Indicator Rig.



A = Bearing rigidly fastened to frame.

B = Point on cross-head.

AC = Oscillating Link.

BC = Short Link

HD = Short Link.

I, K = Bearings on frame.

HN = Long rod.

M, N = Crank and head indicators.

For this reason the length of H D should be $\frac{4\frac{9}{16}}{16\frac{5}{16}} \times 6\frac{3}{8} = 1\frac{25}{32}$ inches

To shorten the length of string as much as possible, the long rod H N was run out past both indicators and thus all the string necessary is a length a little in excess of the circumference of the drum or about 7 inches.

To connect up and disconnect the indicator drums with the rod H N the following scheme was used. O is a U-shaped piece of wrought-iron but around H N and clamped in whatever position desired by the bolt P which passes through both legs of O. Q is a square piece of wood 3" × $\frac{3}{8}$" × $\frac{3}{8}$" to the middle of which is fastened the cord from the indicator. To connect up Q is taken between thumb and forefinger and the string pulled all the way out

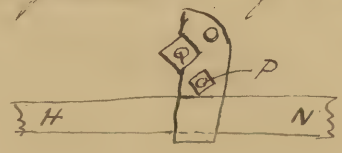


Fig. 2.

when Q can be released to slip into the triangular notch in O.

As a test of the accuracy of the rig, the stroke was divided up into distances of 1" and a vertical line corresponding to each of these divisions was made upon a blank card on the indicator drum, 1/1000th in the limits of measurement each of these divisions was found to be the same.

The principal dimensions of the engine are as follows: —

Stroke	16"
Diam. of Cylinder	8 1/32"
Diam. of Piston Rod	1 1/2"
Diam. of Fly-wheel	4'6"

The engine does not run up to catalogued speed (280 R.P.M.) because the pulley on the governor has been replaced by one of smaller diameter.

The clearance volume was measured by putting the engine on dead center and weighing the amount of water required to fill the space. Packet, rubber was placed under the valves and the latter were clamped down in order to prevent the escape of water into the valve-chests. There was still a slight leak past the piston. This was corrected for by noting the time required to fill the space the first time — call it t seconds. Then $\frac{t}{2}$ seconds after it was full, enough more water was added to fill up the space lost by the leakage. This latter quantity is the amount to be subtracted from the first quantity to give the true amount. The readings are given on the following page.

		Head.		
Wt. Before	Wt. After	Wt. Before correction	Wt. After	Corrected Wt.
5 lbs. $15\frac{3}{8}$ oz	4 lbs. $8\frac{3}{4}$ oz	$13\frac{7}{8}$ oz	$13\frac{1}{8}$ oz	1 lb. $5\frac{7}{8}$ oz
4 – $8\frac{3}{4}$	3 – $1\frac{7}{16}$	$12\frac{1}{4}$	$10\frac{13}{16}$	1 – $5\frac{7}{8}$
7 – $0\frac{13}{16}$	5 – $10\frac{1}{8}$	$10\frac{13}{16}$	$10\frac{1}{4}$	1 – $6\frac{1}{8}$
			Mean	1 – $5\frac{15}{16}$
		Crank		
Wt. Before	Wt. After	Wt. Before correction	Wt. After	Corrected Wt.
7 lb – $4\frac{15}{16}$ oz	5 lbs $13\frac{1}{4}$ oz	$9\frac{1}{8}$ oz	$8\frac{3}{16}$ oz	1 lb $6\frac{3}{4}$ oz
5 – $13\frac{1}{4}$	4 – $5\frac{7}{16}$	$8\frac{3}{16}$	$7\frac{3}{16}$ oz	1 – $6\frac{13}{16}$
4 – $5\frac{7}{16}$	2 – $13\frac{5}{8}$	$7\frac{3}{16}$	$5\frac{15}{16}$	1 – $6\frac{9}{16}$
				1 – $6\frac{11}{16}$

Head Clear. Vol = Vol. of $21\frac{15}{16}$ oz of H_2O = .021938 cu. ft.

Crank " " = " " $22\frac{11}{16}$ " " " = .02269 " "

Mean Clear. Vol = .02231 cu. ft.

% Clearance - Head = $\frac{.02194 \times 1728}{\frac{\pi}{4} \times (8\frac{1}{32})^2 \times 16} = 4.66\%$

% " - Crank = $\frac{.02269 \times 1728}{\frac{\pi}{4}\{(8\frac{1}{32})^2 - (1\frac{1}{2})^2\} \times 16} = 5.01\%$

% Clearance (Mean) = $\frac{4.66 + 5.01}{2} = 4.835\%$

Two Thompson Engraving Indicators were used, one for each end of the cylinder. Two forty pound springs were used in all the tests and these springs were carefully calibrated while hot. In so far the indicator was placed in connection with a vessel to which steam could be admitted The pressure in the vessel and therefore on the indicator piston was measured by means of a steam-gauge which had been previously calibrated. Steam is admitted by opening the valve slightly. As the pointer on the gauge slowly swings around the circle is traced against a blank card on the indicator-drum and just as the pointer passes the 10 lb. mark the card is pulled. The beginning of the almost horizontal line thus obtained is taken as the indicator

estimate that corrections to the corrected gauge pressure. This was done for each ten pounds from 0 to 100 lbs. with the pressure increasing and decreasing. Two complete sets of readings were taken for each spring and they are given on the following pages. It will be seen that the two sets agree very closely. Below is given the calibration of the gauge (#260627)

True	Ascending	Descending
10	10.5	11.0
20	19.8	20.2
30	29.8	30.1
40	39	40.0
50	48.9	49.8
60	58.9	59.9
70	68.3	69.3
80	78.2	79.3
90	88	89.1
100	98.1	98.9

Crank Indicator #325, Ascending.

Pressure	Ordinate	Ordinate	Mean Ord.	Spring Scale
0	0	0		
9.5	.23"	.23"	.23"	41.3
20.2	.50	.50	.50	40.4
30.2	.74	.74	.74	40.8
41.0	1.01	1.00	1.005	40.8
51.1	1.26	1.25	1.255	40.7
61.1	1.52	1.51	1.515	40.3
71.7	1.80	1.79	1.795	40.0
81.8	2.06	2.05	2.055	39.8
92	2.32	2.32	2.32	39.7
			Mean =	40.95

Crank Indicator #325 Descending.				
Pressure.	Ordinate	Ordinate.	Mean Ord.	Spring Scale
90.9	2.33"	2.32"	2.325"	39.2
80.7	2.14	2.16	2.15"	37.6
70.7	1.86	1.88	1.87	37.8
60.1	1.59	1.60	1.595"	37.8
50.2	1.33	1.33	1.33	37.8
40.0	1.05"	1.06	1.055"	37.9
29.9	.79	.79	.79	37.9
19.8	.53	.52	.525"	37.8
9.0	.25"	.25"	.25"	36.0
0	0	0		
			Mean =	37.76

Mean (Ascending and Descending)
= 39.36 lbs. per inch.

Head Indicator #3200, Ascending				
Pressure	Ordinate	Ordinate	Mean Ord	Spring Scale
0	0	0		
9.5	.22"	.22"	.22"	43.3
20.2	.49	.49	.49	41.2
30.2	.72	.72	.72	42.0
41.0	1.00	.99	.995	41.2
51.1	1.24	1.24	1.24	41.2
61.1	1.49	1.49	1.49	41.0
71.7	1.77	1.77	1.77	40.5
81.8	2.02	2.02	2.02	40.5
92.0	—	2.24	2.24	40.2
			Mean =	41.23

Fea á Inaicator #3260 Descending.				
Pressure	Ordinate	Ordinate	Mean Ord.	Spring Scale
90.9	—	2.24"	2.24"	40.6
80.7	2.06	2.07	2.065	39.1
70.7	1.80	1.80	1.80	39.3
60.1	1.53	1.54	1.535	39.2
50.2	1.27	1.28	1.275	39.4
40.0	1.01	1.02	1.015	39.4
29.9	.76	.75	.755	39.6
19.8	.49	.50	.495	40.0
9.0	.22	.23	.225	40.0
0	0	0		
			Mean =	39.62

Mean (Ascending and Descending)
= 40.43 lbs. per inch.

In running the tests the exhaust steam was carried through a condenser, condensed, and afterward weighed. The quality of the steam used was measured by a Barrus Calorimeter placed in the steam main about 18" above the throttle-valve. Indicator cards were taken every 15 minutes and simultaneously the brake and counter were read. The pressure-gauge and upper and lower thermometers of the calorimeter were also read every 15 minutes.

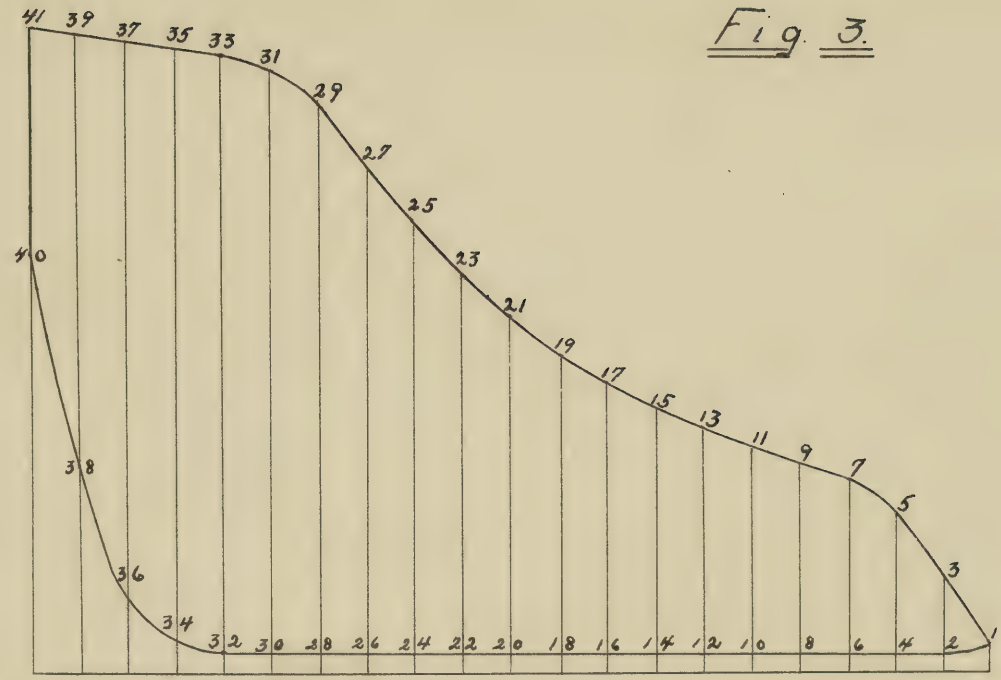
It was attempted to keep the load constant throughout the test by adjusting from time to time the tension of the screws of the Prony brake so that the scale-beam is always balanced at the same point. This was not quite attained owing to variations in pressure and speed. The brake rearings and pulleys

are therefore appended to show to what extent the load varied. The mean speed used in working out the tests was obtained by dividing the total no. of revolutions during the test by the total time.

The entropy-temperature analysis in these tests was based upon mean indicator cards. Blue-prints of these are shown along with the entropy diagrams.

To show the method of obtaining the mean cards the first test is worked out in full. The card was divided up into 21 ordinates numbered as shown on the sketch-card on the following page. Each of these ordinates except those on the straight portion of the exhaust line was scaled off in hundredths of an inch and the means of all the corresponding points on the card

Fig. 3.


41 39 37 35 33 31 29 27 25 23 21 19 17 15 13 11 9 7 5 3 1
40 38 36 34 32 30 28 26 24 22 20 18 16 14 12 10 8 6 4 2

cards were multiplied by the spring-scale of the head indicator (40.4 lbs per inch), the crank cards were then combined in a similar way and the means multiplied by the spring scale of the crank-indicator (39.4 lbs. per inch). Then the mean of these two sets was taken as the ordinate for that point on the mean card. These are also the ordinates of the card in the pressure-volume measurement of the indicator-diagram.

On the following pages are given the values of these ordinates in inches as scaled off from the original cards for Test #1.

-Crank-

Card No.	1	2	3	4	5	6	7	9
1	.11	.05	.32	.05	.37	.05	.39	.44
2	.11	.06	.37	.05	.41	.05	.45	.47
3	.09	.02	.35	.02	.38	.02	.41	.45
4	.07	.01	.34	0	.37	0	.41	.44
5	.09	.04	.32	.04	.36	.04	.39	.42
6	.07	.01	.30	.01	.34	.01	.37	.41
7	.10	.05	.30	.04	.34	.04	.38	.40
8	.11	.06	.34	.05	.37	.05	.41	.44
9	.11	.06	.35	.06	.40	.06	.43	.46
10	.10	.04	.34	.04	.37	.04	.41	.45
11	.08	.04	.33	.04	.37	.04	.40	.43
12	.07	.03	.34	.03	.38	.03	.41	.44
13	.11	.05	.34	.05	.39	.04	.43	.47
14	.09	.05	.34	.05	.37	.05	.41	.45
15	.10	.06	.34	.06	.37	.06	.40	.44
16	.09	.05	.32	.05	.36	.05	.39	.43
17	.11	.06	.35	.06	.38	.06	.41	.45
Mean Ord.	.095	.043	.334	.041	.372	.041	.406	.441
Press. in Lbs.	373	1.69	13.16	1.62	14.66	1.62	1600	17.38

	Crank							
Card No.	11	13	15	17	19	21	23	25
1	47	54	.64	.66	.71	.80	.92	102
2	55	58	59	.72	.78	92	94	122
3	49	54	59	.67	.75	82	95	103
4	49	54	56	.66	71	82	90	106
5	47	51	55	.62	71	77	90	100
6	45	51	55	.63	.69	78	89	99
7	44	49	59	.63	.69	77	90	1.06
8	48	54	.62	.67	.74	83	96	113
9	51	57	59	70	.76	87	96	112
10	48	55	57	.69	.74	86	94	1.01
11	47	52	60	.66	72	83	93	111
12	49	54	57	.67	72	87	93	1.00
13	54	60	68	71	87	93	107	126
14	49	54	58	65	74	82	95	1.05
15	47	53	56	65	72	81	93	1.02
16	46	52	56	65	73	79	93	1.00
17	48	54	59	66	73	82	95	1.04
Mean Ord.	484	539	587	665	731	828	941	1067
Press in Lbs.	1907	2124	2312	2630	2880	3262	3708	4280

-Crank-

Card	26	27	28	29	30	31	32	33
1	.05	124	.05	143	.06	180	.11	195
2	.05	130	.05	166	.06	188	.12	195
3	.02	129	.02	141	.03	185	.07	210
4	0	122	0	148	.01	177	.07	191
5	.04	119	.04	136	.05	164	.08	206
6	.01	121	.01	135	.02	178	.08	192
7	.04	119	.04	131	.05	178	.10	191
8	.05	129	.05	143	.05	189	.11	204
9	.07	134	.07	155	.08	197	.13	210
10	.04	126	.04	152	.05	187	.11	198
11	.04	126	.04	138	.05	180	.10	199
12	.03	126	.03	152	.03	183	.10	199
13	.04	127	.04	153	.05	192	.11	199
14	.05	130	.05	144	.06	184	.12	212
15	.05	124	.05	137	.06	183	.12	200
16	.05	125	.05	136	.07	182	.11	204
17	.06	129	.06	141	.07	181	.11	207
Mean Ord.	.041	1.265	.041	1448	.050	1830	.103	1954
Press. in Lbs	162	4964	162	5713	197	7210	406	7683

Crank

Cara No.	34	35	36	37	38	39	40	41
1	26	199	50	2.03	84	2.02	129	206
2	25	197	47	204	85	203	126	202
3	22	216	46	217	81	220	127	221
4	21	195	43	196	80	197	122	198
5	23	222	48	225	85	227	139	228
6	22	200	45	201	82	203	124	201
7	26	200	47	201	87	204	127	205
8	26	211	49	213	86	215	131	216
9	27	215	53	218	92	219	134	220
10	27	203	52	205	93	206	130	208
11	23	209	46	211	82	213	131	215
12	24	205	46	207	82	205	216	209
13	26	211	49	213	86	216	133	218
14	27	220	56	221	92	223	138	225
15	28	215	52	213	91	215	133	216
16	25	214	46	215	86	216	133	217
17	27	220	49	220	86	221	137	222
Mean Ord.	250	2090	485	2108	859	2119	1306	2136
Press. in lbs.	985	8235	1911	8313	3384	8353	5161	8432

- Head -

Card No.	1	2	3	4	5	6	7	9
1	.06	0	.32	0	.35	0	.39	.41
2	.07	.02	.36	.02	.41	.02	.44	.46
3	.03	-.04	.30	-.04	.34	-.04	.36	.39
4	.05	0	.36	0	.41	0	.43	.46
5	0	-.07	.29	.07	.30	-.07	.33	.36
6	.06	.01	.32	.01	.35	.01	.38	.41
7	.07	.01	.32	0	.35	0	.38	.41
8	.07	.01	.32	-.01	.35	-.01	.38	.44
9	.06	-.02	.32	-.02	.35	-.02	.38	.41
10	.09	.03	.36	.03	.40	.03	.45	.46
11	.04	-.02	.36	-.02	.35	-.02	.39	.42
12	.08	0	.38	0	.41	0	.44	.47
13	.17	.10	.43	.10	.47	.10	.50	.53
14	.16	.07	.40	.07	.44	.07	.47	.50
15	.13	.10	.45	.10	.48	.10	.51	.53
16	.12	.06	.38	.06	.41	.06	.44	.47
17	.08	.02	.35	.02	.38	.02	.41	.45
Mean Ord.	.085	.0153	.351	.0143	.356	.0143	.415	.447
Press. in Lbs.	342	618	1418	578	1438	578	1677	1806
Mean Press from H.P.C.	3.58	1.16	13.67	1.10	14.52	1.10	16.38	17.72

Card No.	Head 11	13	15	17	19	21	23	25
1	.45	.50	.54	.60	.66	.74	.87	.92
2	.50	.55	.60	.66	.72	.83	.92	1.01
3	.42	.47	.51	.56	.63	.71	.82	.90
4	.49	.56	.60	.65	.71	.81	.90	1.03
5	.40	.43	.48	.52	.60	.66	.54	.87
6	.44	.48	.52	.59	.66	.73	.85	.90
7	.44	.48	.57	.58	.63	.71	.84	.89
8	.45	.50	.54	.62	.66	.75	.86	.92
9	.44	.51	.55	.64	.69	.76	.87	.96
10	.49	.53	.60	.67	.72	.81	.92	1.01
11	.45	.50	.55	.61	.67	.75	.86	.96
12	.50	.55	.59	.66	.71	.82	.90	1.00
13	.56	.61	.66	.73	.80	.87	1.00	1.06
14	.54	.59	.63	.69	.77	.85	.96	1.03
15	.59	.63	.68	.76	.82	.91	1.01	1.11
16	.50	.54	.60	.65	.72	.79	.90	.99
17	.48	.53	.57	.63	.72	.79	.91	.98
Mean Ord.	.478	.532	.571	.635	.712	.780	.890	.969
Press. in Lbs	1951	2230	2307	2565	2876	3151	3596	3915
Mean Press. from H+C	1919	2177	2309	2572	2878	3206	3652	4067

Head

Card No.	26	27	28	29	30	31	32	33
1	0	114	0	121	.01	157	08	1.5[illegible]
2	.02	120	.03	137	.05	174	.12	1.5[illegible]
3	.05	106	.05	121	.04	145	04	1.82
4	0	120	0	132	.01	170	.06	1.82
5	-.07	107	-.06	119	-.04	147	.03	1.86
6	-.01	114	.01	119	.01	156	.07	1.80
7	0	108	.01	117	.03	156	.07	1.79
8	-.01	117	0	121	.02	159	.08	1.87
9	-.02	119	-.01	125	.01	167	07	1.91
10	.02	122	.02	131	.05	1.69	.11	1.91
11	-.02	106	-.01	125	0	1.63	.04	1.85
12	0	121	.01	130	.03	1.69	.08	1.90
13	.09	12.8	.10	138	.13	176	.18	203
14	.06	133	.07	136	.09	163	.13	1[illegible]7
15	.10	132	.10	141	.12	181	.17	201
16	.06	116	.06	131	.08	157	.13	193
17	.02	115	.02	132	.04	156	.08	1.71
Mean Ord.	.118	1170	.176	1.280	353	1597	.092	1.885
Press In Lbs.	476	47.27	711	5171	1.42	6464	3.[illegible]0	[illegible]
Mean Press from HC	1.[illegible]	4.845	1.16	.5442	1.70	68.37	.388	[illegible]

Head

Card No.	34	35	36	37	38	39	40	41
1	.27	1.93	.56	1.97	1.02	2.04	1.55	2.05
2	.31	1.94	.57	1.97	1.03	2.00	1.56	2.04
3	.23	1.99	.49	2.04	.98	2.09	1.52	2.11
4	.26	1.91	.53	1.92	1.02	1.97	1.51	1.96
5	.22	2.03	.57	2.08	1.01	2.13	1.61	2.15
6	.26	1.92	.48	1.94	.97	1.96	1.56	2.02
7	.31	1.94	.56	1.95	1.05	2.00	1.59	2.03
8	.30	2.00	.57	2.01	1.06	2.06	1.60	2.07
9	.25	1.99	.53	2.01	1.00	2.06	1.59	2.08
10	.31	1.99	.56	2.04	1.02	2.06	1.60	2.07
11	.23	1.99	.48	2.01	.97	2.06	1.55	2.07
12	.28	2.00	.56	2.04	1.01	2.06	1.56	2.06
13	.36	2.14	.64	2.16	1.11	2.20	1.69	2.21
14	.36	2.15	.62	2.19	1.15	2.24	1.71	2.25
15	.39	2.09	.64	2.12	1.11	2.15	1.65	2.16
16	.32	2.07	.61	2.14	1.06	2.14	1.65	2.15
17	.25	2.06	.52	2.14	1.00	2.17	1.64	2.18
Mean Ord.	290	2065	543	2043	1033	2082	1590	2098
Press in lbs.	1172	8120	2194	8242	4161	8403	6424	8484
Mean Press from Hd	1078	8177	2050	8277	3772	8378	5792	8458

The mean card was then measured as shown in the blue-print, the scales being 1" = 25 lbs and 1" = 114 cu. ft. = $\frac{4}{15}$ foot. The area of the card is found to be 7.107 sq. in. by the planimeter.

The area scale = $25 \times \frac{4}{15} = 6.67$ ft. lbs. per sq. in.

and I.H.P. = $\frac{6.67 \times \text{Area card} \times \text{Mean area piston} \times \text{No strokes}}{33000}$

The no. strokes see page 38

$= \frac{(4186 - 2710) \times 2}{4 \times 60} = 327.52$ strokes

or 163.76 revs. per min.

The mean piston area

$= \frac{\pi}{4} \times (8\frac{1}{32})^2 - \frac{\frac{\pi}{4} \times (1\frac{1}{2})^2}{2} = 50.66 - .88$

$= 49.78$ sq. in.

Finally

the I.H.P. = $\frac{6.67 \times 7.107 \times 49.78 \times 327.52}{33000}$

$= 23.43$ H.P.

Moisture in steam.

The pressure by gauge =

91.53 lbs $+ 14.61 = 106.14$ lbs. absolute.

But the gauge (#210013, see page 34) reads 2.29 lbs. high so that the corrected pressure = 103.85 lbs. absolute which is equivalent to a temperature of 330.33° F.
T_1 was found to be 324.0° F so that the radiation amounts to 6.33° and the temperature of the steam escaping from the calorimeter is 259° + 6.33° = 265.33° or the escaping steam is 53.65° superheated (= 265.33° − 211.68°)

But from the formula

$$x = \frac{q_a + r_a + .48(T_{sup} - T_{sat}) - q_p}{r_p}$$

$$= \frac{185.4 + 966.1 + .48 \times 53.65 - 300.8}{881.9}$$

$$= .989$$

or % Moisture = 1.1%

Water Consumption.

The total water consumption (see page 38) was 4827.5 lbs. and steam used per stroke = $\frac{4827.5}{39115 \times 2}$ = .06165 pounds.

But this should be corrected for the amount of moisture in the steam by multiplying by the factor $\frac{g+xr}{g+r}$ and the corrected water consumption

$= .0165 \times \frac{289.4+.989\times890}{289.4+890} = .06113$ lbs. per stroke

and the number of strokes per lb. of cylinder feed $= \frac{1}{.06113} = 16.36$ strokes

The volume of piston displacement per lb. of cylinder feed = 7.07 cu. ft. per stroke × 16.36 strokes

$= \frac{49.78\times16}{1728} \times 16.36 = 7.541$ cu. ft.

The work per lb. of cylinder feed is therefore $7.107 \times 6.07 \times 49.78 \times 16.36$

$= 38,400$ ft. lbs.

The steam consumption per horse-power per hour $= \frac{.06113\times60\times327.52}{23.42} = 5.129$ lbs. per H.P.

The weight of cushion steam was not found by choosing some point on the compression curve when the exhaust valves are known to be closed and

Multiply the pressure and volume at that point. The point chosen for Test # was point 34 and the pressure was 10.78 lbs. gauge = 25.39 lbs. absolute. As this is below the pressure of 26.9 lbs. (See Reeve's Entropy - Temperature Analysis page 15) the weight thus found must be corrected for moisture by dividing by the x given on the entropy diagram corresponding to 25.4 lbs. This is found to be .997 and the weight of moisture steam is therefore = $\frac{\text{vol.}}{\text{sp. vol.}} \times \frac{\text{No. strokes}}{.997}$

$= .\frac{0915}{5.13} \times \frac{16.36}{.997} = .0941$ lb.

since the vol = Clearance vol + $\frac{3}{20}$ Cylinder volume

$= .02231 + \frac{3}{20} \times \frac{49.78 \times 16}{1728} = .0915$ cu ft.

The Clearance volume per pound of cylinder fluid = clearance volume per stroke X no. of strokes = .02231 X 16.36 = .365 cu. ft.

We now have all the data necessary to lay down the card in the pressure-volume quadrant of the entropy diagram

(as outlined in Reeve). After that has been done the entropy - temperature diagram is found and the various areas upon it are measured with the planimeter. These areas should then be multiplied by the proper scale (1 sq. in. = 4481.28 ft. lbs) to get the results in the form given in the tables on the right of each diagram; but in our case we found that the paper had shrunk about 1 part in 201 in area so that the proper scale by which to multiply the areas is $4481.28 \times (1.005)^2$

$= 4481.28 \times 1.01 = 4526.1$ (ft-lbs = 1 inch)

The duration of the various tests (see pages 37-59) varied. Tests #1 and #2 were four hours each. But these were with light loads. With the heavier loads, the condensing water, since it was used over and over again,

got too hot to do its work properly and we were forced to stop at the end of two hours and in one case (Test # 3) at the end of one hour and three-quarters.

Two gauges were used one - #210013 for Test #1 and the other #260267 for Tests #2 to #6. Their calibrations are given on the following pages. #260267 was tested from time to time and practically no variation was detected. It is a very satisfactory gauge.

Calibration- Gauge #210013			
True	Asc.	Desc.	Correction
0	.6	.8	-.7
10	11.8	13.0	-2.4
20	22.0	23.2	-2.6
30	31.8	33.2	-2.5
40	42.0	43.4	-2.6
50	52.0	53.2	-2.6
60	62.2	63.2	-2.7
70	72.0	73.0	-2.5
80	82.0	82.8	-2.4
90	92.0	92.2	-2.1
100	102.0	102.2	-2.1
		Mean =	-2.29

Calibration – Gauge #260267.			
True	Asc.	Desc.	Correct.
55	53.7	54.5	+.9
60	59.0	59.9	.6
65	63.5	64.7	.9
70	69.0	69.9	.6
75	74.0	74.9	.6
80	79.2	80.0	.4
85	84.4	85.0	.3
90	89.2	89.8	.5
95	94.0	94.7	.7
100	99.0	99.7	.7
105	103.9	104.4	.8
		Mean=	+.64

Observations and Results

Test #1

Time	T_1	T_2	Pressure	Brake
10-42	324	259	89	127
-57	324	258	88	130
11-12	327	257	94	130
-27	329	259	87	130
-42	330	267	97	130
-57	324	265	87	130
12-12	319	256	89	124
-27	322	262	90	127
-46	322	256	92	130
-57	321	258	89	127
1-12	322	258	91	129
-27	320	258	89	130
-42	322	256	92	130
-57	325	261	95	130
2-12	324	258	89	129
-27	325	256	93	128
-38	328	259	95	130
Sums	6508	4403	1556	2181
Means	3240	259.0	91.53	128.9

Test #1

Time	R.P.M.	Time	wt full	Wt. empty	Net Wt
10-40	164	10-40	666	168	498
-55	163	11-09	656.5	171	485.5
11-10	162	-36	673	175.5	497.5
-25	163	12-05	665.5	172	493.5
-40	165	-33	682	172.5	509.5
-55	164	1-02	672	174	498
12-10	163	-31	671.5	168	503.5
-25	164	-46	715	169.5	545.5
-40	164	2-14	689	175	514
-55	163	2-40	437	154.5	282.5
1-10	170	Totals	6527.5	1700	4827.5
-25	157				
-40	165				
-55	166				
2-10	168				
-35	163				
-37	160				
Mean	1643				

Barometer 29.770"

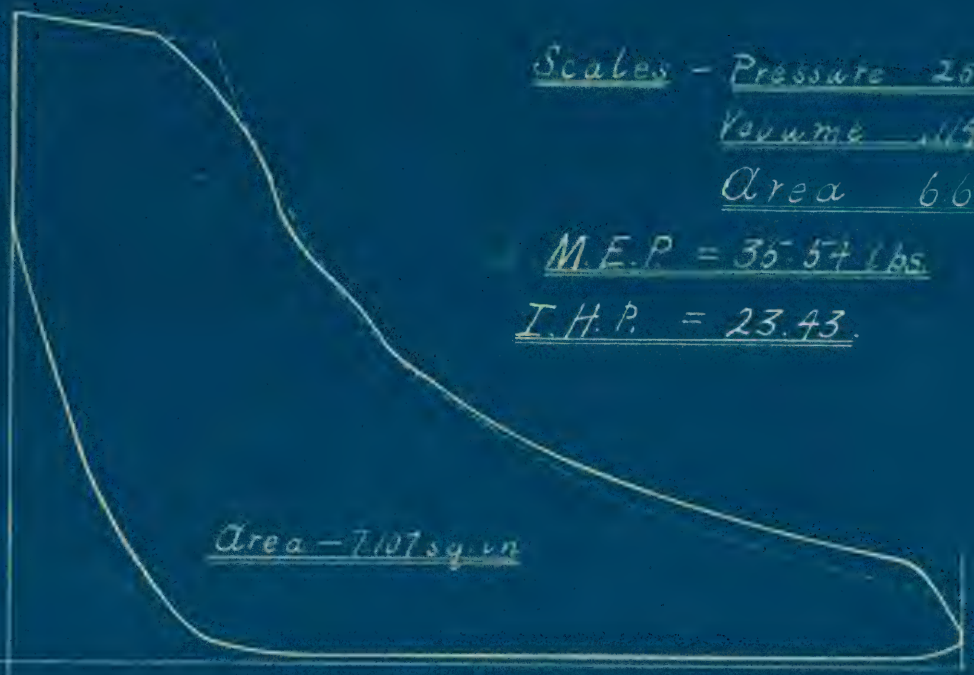
Counter - start 2710

- stop 41861

Total Revs. 39151

Duration 4 hrs.

Mean Card – Test #1



Scales – Pressure 25 lbs = 1"

Volume .114 cu. ft = 1"

Area 6.67 ft. lbs. = 1"

M.E.P = 35.54 lbs.

I.H.P. = 23.43.

Area – 7.107 sq. in

Entropy temperature Diagram blank for Steam engines.

EXPLANATORY PAMPHLET PUBLISHED SEPARATELY

N.B.

Test #1
8×16 Porter-Allen
April 2, 1896
Cylinder Unjacketed.
77% Moisture
50 Lbs
160°F
Blk

One

107500
39500
37951
716
67669
805
5010
5.50
12.60
1070
68490

LIBRARIAN OF CONGRESS

Test #2

Time	T_1	T_2	Pressure	Brake
10-17	324	263	88	114
-32	322	263	85	114
-49	322	261	86	114.5
11-01	323	263	87	113.5
-16	321	263	82	113.5
-30	321	260	82	113
-46	327	262	91	114
12-01	322	264	82.5	111
-17	320	259	81.5	115.5
-31	323	259	86	117
-46	330	264	95	114
1-01	326	264	88.5	113
-16	324	263	85.5	114
-31	326	262	89.5	112
-46	326	265	87.5	115
2-02	323	262	84	117
-11	326	262	90	113
Sums	5506	4559	1471.0	1938
Means	323.9	262.3	86.53	114.0

Test #2

Time	RPM	Time	Wt. full	Wt empty	Net Wt
10.16	163	10-40	692	161	531
-30	165	11-05	6555	183	472.5
45	16.6	-35	7075	1645	543
11-00	165	12-05	711	162	549
-15	164	-35	706	160.5	545.5
-30	165	-50	704.5	166	5383
45	164	1-10	724	160	564
12-00	163	-40	723	166.5	555.5
-15	163	2-02	705	162.5	542.5
-30	164	-15	527	157	370
-45	164	Totals	68545	1643	5211.5
1-00	165				
-15	165				
-30	165				
-45	164				
2-00	163				
-10	164				
Mean	1642				

Duration- 4hrs

Barometer- 29.764"

Counter- start 80055

stop 119072

Counter stopped

1.15- 9533

1.30- 11514

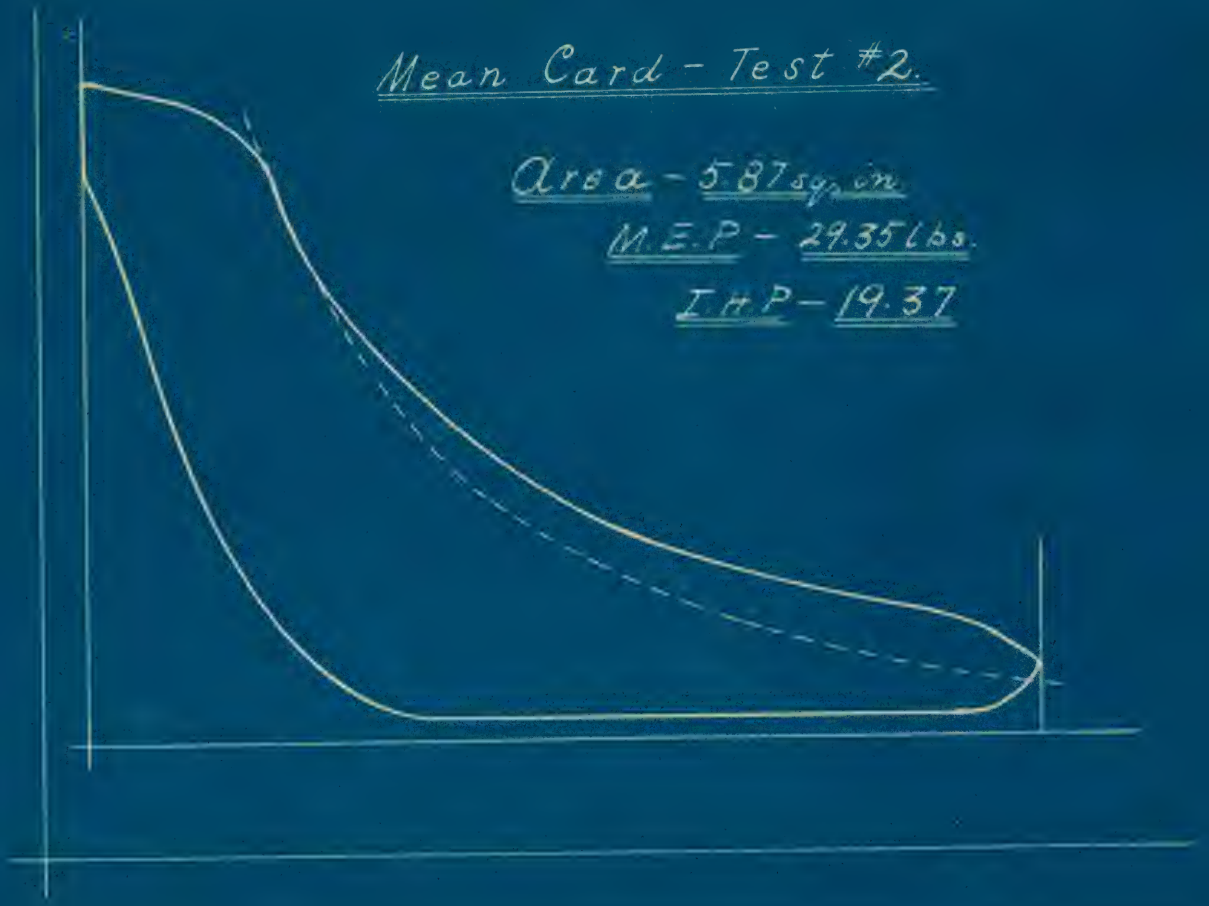
Total Revs (in 3¾ hrs) 37036

Mean Card - Test #2.

Area - 5.87 sq. in.

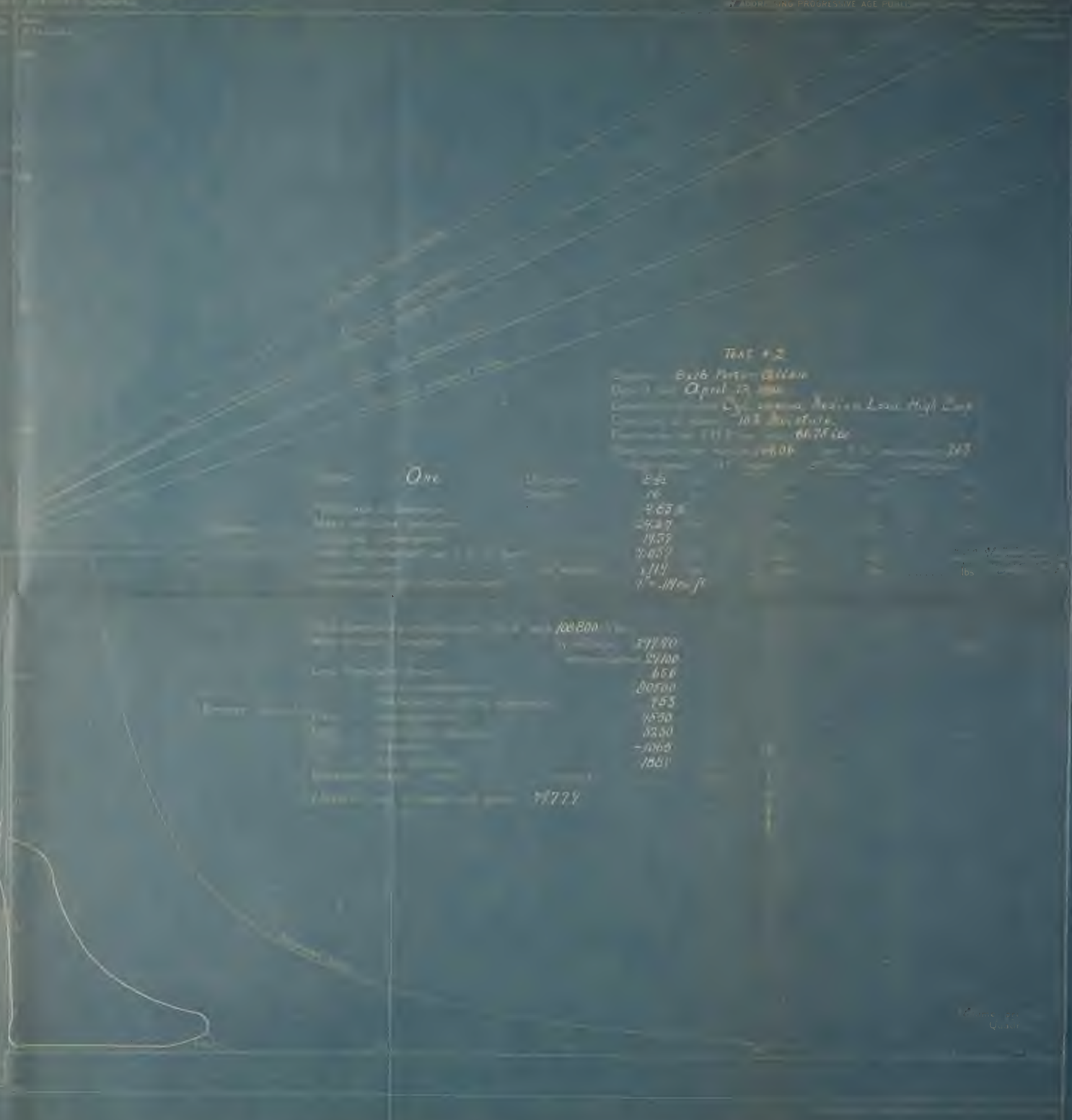
M.E.P - 29.35 lbs.

I.H.P - 19.37



Curves showing the Horsepower required for Steam Vessels

ENTERED ACCORDING TO ACT OF CONGRESS
BY PROGRESSIVE AGE PUBLISHING



Test #3

Time	T_1	T_2	Pressure	Brake
2-32	314	220	72.5	132.5
-47	311	219	69	140
3-02	314	221	59	131
-17	308	216	66	130
-31	309	219	69.5	135
-46	306	220	66.5	135
4-02	308	220	68	133
4-11	302	221	60.5	130
Sums	2462	1756	531.0	1070.5
Means	307.75	219.5	66.38	133.8

Test #3

Time	R.P.M.	Time	Wt. full	Wt. empty	Net Wt
2-30	156	2-55	702	156	546
45	156	3-15	654	164	490
3-00	157	-35	687	171	516
-15	161	-55	655	158	497
-30	159	4-15	696	165	531
-45	158	Totals	3394	814	2580
4-00	157				
4-10					
Mean	158.0				

Barometer- 29.6.00"

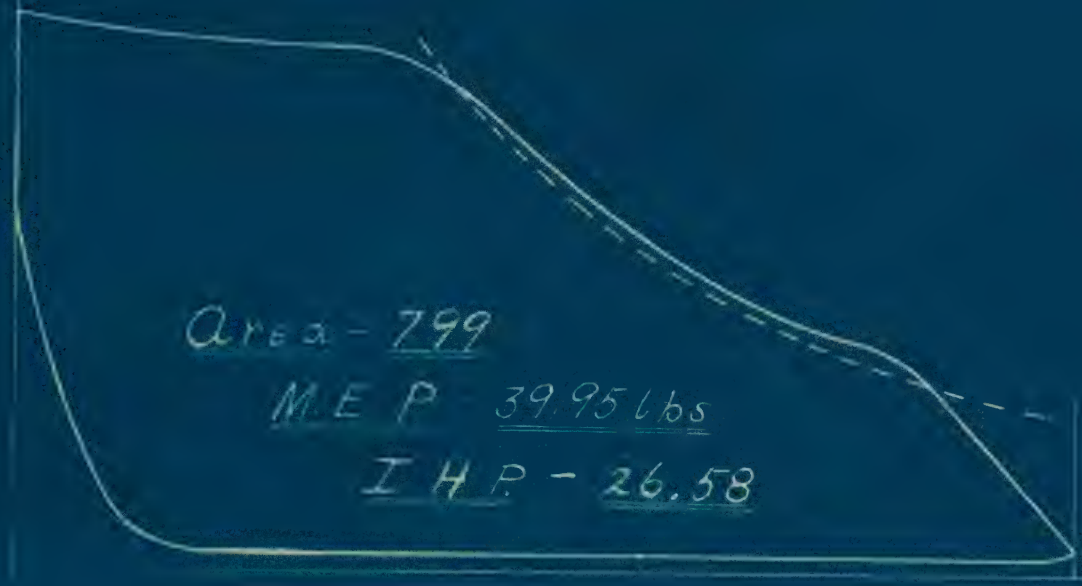
Counter- start 75639

stop 92229

Total Revs. 16590

Duration- 1 3/4 hrs.

Mean Card – Test #3



Area – 799

M E P 39.95 lbs

I H P – 26.58

Entropy-temperature Diagram blank for Steam engines.

Arranged by Sidney A. Reeve, M.E.

EXPLANATORY PAMPHLET PUBLISHED SEPARATELY

Entropy-temperature Quadrant

Entropy Scale

Area contained in this quadrant

1 square inch = 2.76 B.t.u. at 0 lbs. P.S.I.

1 square inch = 2.76 B.t.u. at 15 lbs.

N.B. ALL SCALES FOR ABOVE CORRECT FROM 0 TO 1 INCH WORK TO SCALES ON MAP

Saturated steam

Water

Temperature-Entropy Quadrant

Entropy scale

Saturated steam

Test *3

Engine 8x16 Porter-Allen

Date of test April 28, 1896

General conditions Cyl. covered, Heavy Load on Lake Comp.

Condition of steam 175 lbs. Gauge

Feed water per I.H.P. per hour 35.0 lbs

Revolutions per minute 183.0 per I.H. card-engine 183

	High-press.	I.P. cylinder	2nd I.P. cylinder	Low-press.
Number	One			
Diameter	8½" in.	in.	in.	
Stroke	16 in.	in.	in.	
Percentage of clearance	4.85 %			
Mean effective pressure	59.93 lbs.	lbs.	lbs.	
Indicated horse power	206.8			
Piston displacement per 1 lb. of feed	6.026 cu. ft.			
Cushion steam by weight	.0459 lbs.			
Volume-scale of indicator-card	1" = 1.14 cu. ft.			

Scale, 4 = 1 ft.

Work theoretically available per 1 lb. of feed 94960 ft-lbs.

Work actually developed	by indicator	34800
	" entropy-diagram	34700
Loss from wire drawing		940
" initial condensation		51400
" cylinder-condensation during expansion		0
Gain " reevaporation		1490
Loss " incomplete expansion		8150
" " clearance		-115
" " back pressure		1860
Percentage-losses from		found / from

Adiabatic curve of heating and gains 60643 ft lbs

ENTERED ACCORDING TO ACT OF CONGRESS IN THE YEAR 1897
BY PROGRESSIVE AGE PUBLISHING COMPANY IN THE OFFICE OF THE LIBRARIAN OF CONGRESS.

Published by Progr

Test #4

Time	T_1	T_2	Pressure	Brake
10-17	324	262	88	79.5
-32	325	259	90	79.5
-46	324	259	89	80
11-01	326	258	92	80
-16	328	256	94	80
-31	326	260	88	80
-46	322	258	83	80
12-01	328	258	91	81.5
-16	320	258	82	79
-31	322	253	89	79
-46	328	260	93	80
1-01	322	258	86	79.5
1-12	327	258	94	84
Sums	4222	3357	1159	1042
Means	324.77	258.23	89.15	80.15

UNIVERSITY OF PENNSYLVANIA

Test #4

Time	R.P.M.	Time	Wt. full	Wt. empty	Net Wt.
10-17	170	10-42	653.5	161	491.5
-30	171	11-12	665	154.5	510.5
-45	169	-39	670.5	163	507.5
11-00	171	12-07	685	165	520
-15	171	36	670	166.5	503.5
-30	169	1-06	654	167	487
-45	169	-15	296	188	108
12-00	172	-15	322.5	152	170.5
-15	172	Totals	46165	1317	32995
30	170				
-45	171				
1-00	170				
1-10	170				
Mean	17038				

Barometer- 29.596"

Counter start 26871, stop 56985

Counter stopped 10.30 - 29375

10.45 - 31468

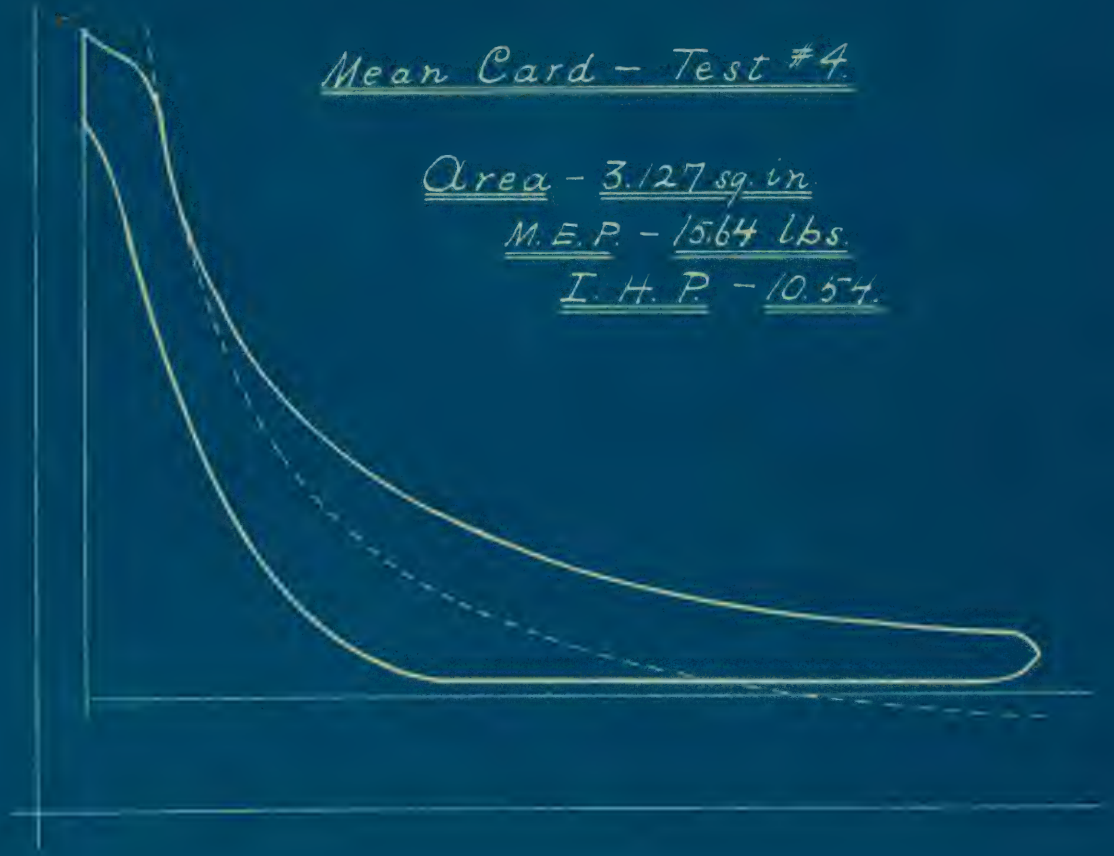
Duration 3hrs Total Revs (in 2¾ hrs) 28021

Mean Card – Test #4.

Area – 3.127 sq. in.

M.E.P. – 15.64 lbs.

I.H.P. – 10.54.



Entropy temperature Diagram blank for Steam engines.

PUBLISHED SEPARATELY

Test # 4

Engine 8x11 Porter-Allen

Date of test April 30, 1898.

General conditions Cyl. Covered, Light load a. High Comp

Condition of steam 41% Moisture

Fuel water per 1 H P per hour 10.7

Revolutions per minute 160.62 per 1 lb fuel-water 9.31

	High press.	1st interm.	2nd interm.	Low press.
Number	One			
Diameter	8½ in.	in.	in.	in.
Stroke	16 in.	in.	in.	in.
Percentage of clearance	4.08 %			
Mean effective pressure	15.60 lbs.	lbs.	lbs.	lbs.
Indicated horse power	10.54			
Piston displacement per 1 lb of fuel	8.088 cu ft			
Cut-off - stroke	.1506 in.	in.	in.	in.
Volume - scale of indicator-card	1" = 114 cu ft			

Work theoretically available per 1 lb of fuel 109000 ft lbs

Was actually developed		by indicator	14300
		by entropy diagram	9630
Loss	from wire drawing		200
	initial condensation		98000
	condensation during expansion		0
Gain	re-evaporation		11400
Loss	incomplete expansion		2160
	clearance		-660
	back-pressure		1750
Receiver - losses	first	second	third

Algebraic sum of losses and gains 90050

Entropy temperature Ignorance

Entropy Scale

N.B

Temperature Scale

Volume Scale

Published by Progressive Age

Test #5

Time	T_1	T_2	Pressure	Brake
10-17	319	265	80	148
10-32	321	263	84	150
10-46	320	264	85	149
11-01	321	263	83.5	149
11-16	323	263	85	150
11-31	318	262	79	145
1146	323	262	85	149
12-01	324	264	86	150
12-11	319	262	80	149
Sums	2887	2368	747.5	1339
Means	320.88	263.11	83.06	148.7

Test #5

Time	R.P.M.	Time	Wt. full.	Wt. empty	Net. Wt.
10-16	167	10.30	666	150.5	515.5
-31	167	-56	658	166.5	491.5
-44	165	11-17	664	168	496
11-00	167	-34	670	182	488
-14	166	-51	669	160	509
-30	164	12-15	650	172	478
44	167	-15	488.5	148	340.5
-59	167	Totals	4465.5	1147	3318.5
12-10	167				
Mean	16633				

Barometer- 29.956"

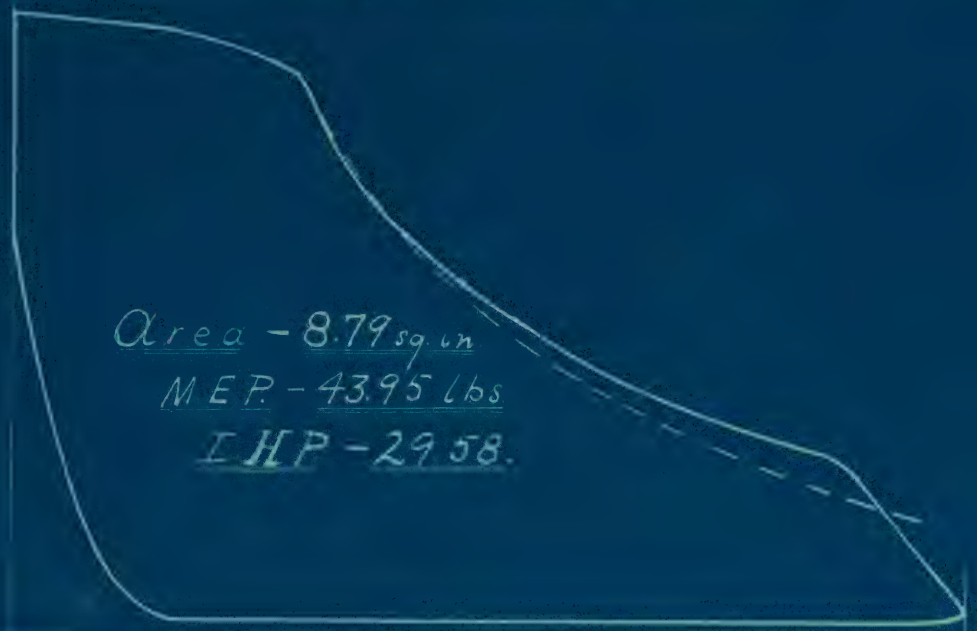
Counter- start (10.30) 28277

Stop 45849

Total Revs 17572

Duration 2hrs.

Mean Card – Test #5



Area – 8.79 sq. in

M.E.P. – 43.95 lbs

I.H.P. – 29.58.

Enthoux temperature Diagram blank for Steam engines.

N B

Water

Test #5

Engine B & W Porter-Allen

Date of test May 19 1898

General conditions Cyl Covered, Heavy Load, Late Camp

Condition of steam Dry Superheated

Temperature per [illegible] 5800 lbs

Evaporation per minute 167.32 per 1 lb feed-water 6.09

	High press	2d Interm	Low press
Number	One		
Diameter	8.6		
Stroke	16		
Percentage of clearance	4.65 %		
Mean effective pressure	43.96 lbs		
Indicated horse-power	29.80		
Piston-displacement per [illegible] of feed	5614 cu ft		
Cushion-steam per weight	6809		
Volume-scale of indicator-card	1" = .0600 cu ft		

Work theoretically available per 1 lb of feed 106.528 ft lbs

Work actually developed	by indicator	35530
	entropy-diagram	34900
Loss from wire drawing		949
	initial condensation	69700
	" condensation during expansion	1033
Gain	re-evaporation	0960
Loss	incomplete expansion	5170
	clearance	126
	back pressure	2055

Receiver-losses, first third

Algebraic sum of losses and gains 71962

Entropy

Scale, 1 = 0.1

ENTERED ACCORDING TO ACT OF CONGRESS IN THE YEAR 1897
BY PROGRESSIVE AGE PUBLISHING COMPANY IN THE OFFICE OF THE LIBRARIAN OF CONGRESS.

Test #3

Time	T_1	T_2	Pressure	Brake
10-02	319	258	81	144
10-17	321	258	84.5	151
10-32	314	257	74	153
10-47	316	257	76.5	150
11-01	318	255	82.5	152
11-16	324	259	92	150
11-31	320	259	84.5	149
11-46	330	262	102	148
11-56	326	264	94	152
Sums	2888	2329	771	1349
Means	320.88	258.78	85.67	149.9

Test #6.

Time	R.P.M.	Time	Wt full	Wt empty	Net Wt.
10-02	168	10-20	709	163	546
-15	169	-40	658	163	495
-30	164	11-00	701	155	546
-45	168	-17	642	158	484
11-00	164	-35	655	164	491
-15	169	-47	535	176	359
-30	168	12-00	563	155	407
-45	169	Totals	4462	1134	3328
-55	—				
Mean	16738				

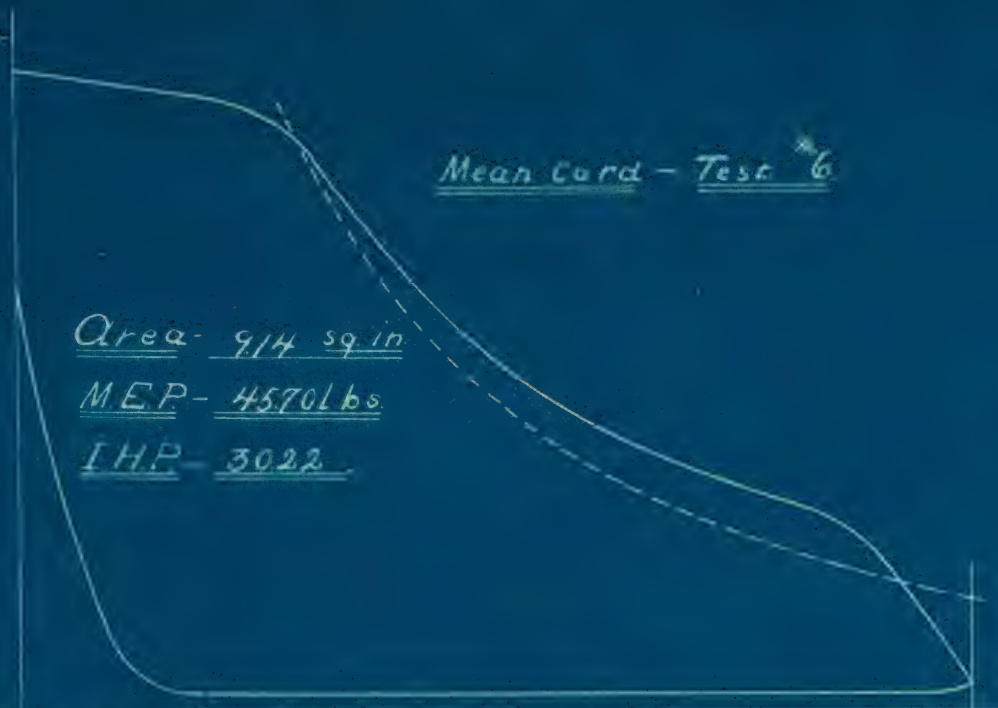
Duration 2hrs.

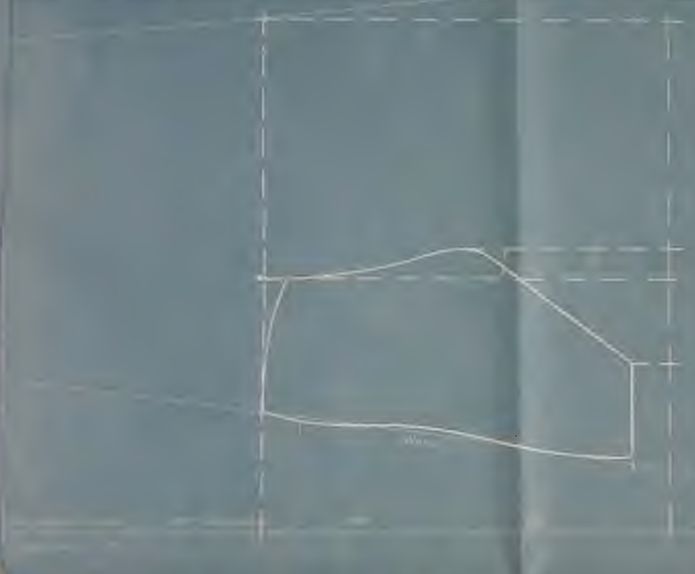
Barometer- 29.740"

Counter start 50709

stop 70478

Total Revs. 19769


Mean Card - Test #6
Area - 9.14 sq in
M.E.P. - 45.70 lbs
I.H.P. - 3022



Test #6
Dale Miller-Allen
May 27 1938
Cut bank Heavy Load, Light Comp
10 % Moisture
5466 lbs
16471 per 1 lb feed-water 5.98

One

# b	
#	
4833	
457	
30.22	
5.518	
.0376	
1"=44 = ft	
	68200
	36250
	35300
	633
	66900
	0
	3780
	8280
	248
	2350
	72730

On the following pages is given a resumé of the results obtained.

The general conditions governing the the Tests are as follows -

It was attempted to keep the steam-pressure between 80 and 85 lbs (except in Test #3) because the maximum ordinate obtainable on the indicator was 2¼" which with a 40 lbs spring is equivalent to 90 lbs. pressure.

In the Porter-Allen engine the admission valves are entirely under control of the governor and, once set, they were not changed throughout the tests. Different cut-off's were obtained by varying the loads.

The exhaust valves were set in three different positions. In Test #1 they are set for moderate compression, in Tests #2 and #4 for earlier or higher

Test No.	1	2	3	4	5	6
Moisture in steam	1.1%	1.0%	2.7%	1.1%	1.0%	1.0%
Feed water per I.H.P. hr	51.31 lbs	6675	5557	1037	5588	5465
R.P.M.	163.76	164.06	158.0	169.82	167.32	164.74
Revs. per lb. of cyl. feed	8.18	7.63	6.56	9.31	6.09	5.98
M.E.P.	35.54	29.27	39.95	15.60	43.95	45.7
I.H.P.	23.43	19.37	25.98	10.54	29.58	30.22
Piston displacement per lb. of cyl. feed	7.54 cu ft.	7.04	6.06	8.59	5.61	5.52
Wt. of cushion st.	.0941	.119	.0489	.151	.0504	.0516
Work— theoretically available	ft. lbs. 107500	108800	94900	109000	106500	108200
— act. developed (indicator)	38500	29745	34800	19300	35530	36250
— act. developed (entropy diag)	37850	29100	34700	19600	34900	35500
Losses— wire-draw.	716	656	996	200	949	633
— initial cond.	67500	80500	51400	98000	64900	66400
— cond. dur. exp.	805	453	0	0	1333	0
Gain— reevap. dur. exp.	5010	7550	1450	11400	5400	5180
Losses— in comp. exp.	5150	5250	8150	2160	8190	8280
— clearance	-1070	-1065	-113	-660	-45	+248
— back press	1260	1880	1860	1750	2055	2350
Sum of losses & gains	68990	74779	60843	90050	71752	72730
Efficiency	35.3%	26.8%	36.6%	18.0%	30.3%	32.8%

Test No.	1	2	3	4	5	6
Work— theoretically available	100%	100%	100%	100%	100%	100%
— act. developed	353%	268%	366%	18%	305%	328%
Losses— wire draw.	.667%	.604%	1.05%	.183%	.892%	.585%
— initial cond.	62.8%	74.2%	54.2%	90.0%	61.0%	61.4%
— cond. dur. exp.	.75%	.417%	0	0	1.25%	0
Gain— reevap. dur. exp.	4.67%	6.45%	1.53%	10.5%	5.07%	4.79%
Losses— incomp. exp.	4.80%	4.83%	8.60%	1.98%	7.7%	7.65%
— clearance	-.99%	-.98%	-1.19%	-.607%	-.042%	+.23%
— back press.	1.17%	1.73%	1.97%	1.61%	1.93%	2.18%
Sum of losses & gains	64.2%	68.7%	64.1%	82.6%	67.6%	67.3%

clearance; in Tests #3 #4 & #5 for both low compression. This last section was found by Trial from the other tests. And is the position in which the loss from clearance volume is 0 or practically 0. see diagrams for Tests #3, 5, 6)

Tests #1 and #6 were run with the cylinder bare of all covering. In Tests #2 — #5 the cylinder was covered with two inches or more of magnesia covering put on in the form of a paste and securely wired in position.

Before running Test #1 it was attempted to get the steam valves as tight as possible by tightening up the pressure-screws, but during the test it was noticed that the governor was not working properly owing to excessive valve-friction.

Before starting Test #2 the pressure-blocks were eased off until the governor worked

injury, with the result that mileage and consequently water consumption were increased. For this reason Test #1 and Test #2 cannot be directly compared with each other for the purpose of showing the effect of covering the cylinder. In Tests #1, #3, and #4 the pressure-points were unchanged but in #5 and #6 they were tightened up as much as the governor would allow.

Below is a statement of the figured conditions obtaining during each test in a tabular form.

Test #1 — Medium load, medium compression, cylinder bare, pressure-plates moderately tight.

Test #2 - Medium load, high compression, cylinder covered, pressure-plates eased.

Test #3 — Heavy load, low compression, low steam-pressure, cylinder covered,

Pressure-plates eased.

Test #4 – Light Load, High Compression, Cylinder covered, Pressure-plates eased.

Test #5 – Heavy Load, Low Compression Cylinder covered, Pressure-plates tight.

Test #6 – Heavy Load, Low Compression, Cylinder bare, Pressure-plates tight.

Conclusions.

The tests show an excessive amount of initial condensation. This loss varies from 54.2% to 93% of the total available energy or from 14.8% to 55% of the indicated work. This is enormous and can only be accounted for on the basis of leakage. Unfortunately the entropy-temperature diagram does not indicate the leakage loss from the other losses. Leakage affects all the others, for this reason. Leakage causes larger steam consumption which in turn causes the moisture in the cylinder-volume quadrant to be shortened up, i.e. the scale of abscissae is less than it should be. But the weight of the entropy diagram is determined by the pressure of

the point on the P-V diagram. Therefore the contents of the card makes the ordinates of the entropy-diagram shorter and increases the apparent initial condensation and some of the other losses.

Recently the engine was shown to leaks by turning it over until the valve was known to be closed. On opening the indicator-cocks steam escaped with great violence. The rectangular dry barbola, shown in a dotted line on the mean cards ~~xxxxx~~ also indicates leakage.

The apparently large gain by re-evaporation which varies from .53% to 6.95% of the total energy = 4.2% to 26% of the work also points to leakage. If the steam-valves leak it means that steam and therefore entropy is being added during

expansion and this would appear in the entropy diagram as an increased heat loss to evaporation.

The loss from condensation during expansion is so much diminished as to disappear in some cases. This shows that enough steam leaks through during the first part of the stroke to take the place of that condensed.

The loss from wire-drawing is low in all the tests but one it was less than 1%. This speaks well for the valve-motion.

The losses from incomplete expansion and back pressure are not very excessive and are unavoidable.

The loss due to clearance volume is low for Tests #1, #2 and #4 and is practically zero for #3 and #5

A comparison of Tests #2, #3 and #4 shows that maximum efficiency corresponds with maximum load. Of course, if the conditions were changed i.e. if the steam pressure were increased and leakage diminished this conclusion might be modified. This result evidently depends upon the fact that the initial condensation is nearly constant for a given pressure and independent of the load; or in other words it takes an almost constant number of heat units to bring the cylinder up to the temperature of the entering steam. This is substantiated by the following comparison of initial condensation per stroke (see page 62)

(Test #2) = $\frac{80500}{15.27}$ = 5260 ~~B.T.U.~~ lbs. per stroke

(Test #4) = $\frac{98000}{18.62}$ = 5260 " " " "

(Test #3) = $\frac{51400}{13.12}$ = 3940 " " " "

But Test #3 was run at a lower pressure

than the other two and therefore the last figure should be corrected. If it be assumed that the initial condensation is proportional to the difference in temperature between the entering steam and the cylinder walls — which seems to be a fair assumption — it becomes for Test #3 (see Entropy diagrams Pages 48, 52)

$$\frac{329 - 220}{312 - 220} \times 3940 = 4630 \text{ B.T.U.'s per lb.}$$

which does not differ greatly from Tests #2 & #4

A comparison of Tests #3 and #5 ought to show the effect of varying the pressure. These show that the efficiency was greater for the lower pressure. But this is not so irrational, because with lower pressures the condensation is less and the work is also diminished. The latter point is largely mitigated by the small gain in re-evaporation and by the fact that the expansion line is close to the hyperbolic (see Test #3).

Finally we have to consider the effect of superheat (Tests #5 and #6). This has practically no effect on the efficiency; as a matter of fact it shows an increased efficiency of two per cent in favor of the bare cylinder. But there are several circumstances small in themselves which taken together explain this:

1. In test #6 the load was heavier.

2. " " " " day was warmer and engine room hotter

3. The two observers were changed around in the two tests and their personal errors enter oppositely.

Therefore, judging from these two tests, under the given conditions of excessive leakage it makes no difference in the efficiency and water consumption whether the cylinder is covered or not.

3 1198 03072 7890
N/1198/03072/7890X

3 1198 03072 7890
N/1198/03072/7890X